

NPS ARCHIVE

1969

JONES, D.

DEVELOPMENT OF A TEST PROCEDURE FOR
SPUTTERED GOLD AS A SOLID FILM GEAR
TOOTH LUBRICANT

by

Dennis Richard Jones

UNITED STATES NAVAL POSTGRADUATE SCHOOL



THESIS

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DEVELOPMENT OF A TEST PROCEDURE

FOR SPUTTERED GOLD AS A SOLID

FILM GEAR TOOTH LUBRICANT

by

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Lieutenant, United States Navy
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Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

NAVAL POSTGRADUATE SCHOOL
April 1969

NPS ARCHIVE
1969
JONES, D.

Thesis J695-61

ABSTRACT

Further work on a solid film lubricant testing apparatus designed and built by M. M. Sampsel was carried out. A power supply was constructed and a sputtered gold film applied to various portions of the apparatus. A torsion bridge was installed and calibrated to determine tooth contact forces. Calibration of strain gages installed to determine tooth radial forces was attempted but difficulties were encountered in determining a strain reading for the unstressed condition. Based on this work, recommendations for design changes were made.

TABLE OF CONTENTS

Section	Page
1. Introduction	7
2. Developing the Power Supply	8
3. Sputtering Processes	15
4. Calibration of Strain Gages to Measure Contact Forces	18
5. Calibration of Strain Gages to Measure Radial Force on the Gear Teeth	26
6. Suggestions for Re-design	31
7. Conclusions and Recommendations	35
Bibliography	37
 Appendix	
I. Wiring Diagram of Pulse and Switching Relays	39
II. Equipment Data	40
III. Assembly Directions	42
IV. Error Analysis for Measurement of Contact Forces	45

LIST OF FIGURES

Figure		Page
1.	Block Diagram of the Stepper Motor Power Supply System	11
2.	Torsional Lever Arm	21
3.	Strain vs. Torque with the Test Gear a Tight Fit on the Test Gear Shaft	22
4.	Strain vs. Torque for a RC 2 Fit Between Test Gear and Shaft	24
5.	Test Set-up for Calibration of the Torsional Strain Bridge	25
6.	Representation of Strain as a Function of Tangential Force and Point of Contact	27
7.	Test Set-up for Calibration of Strain Gages to Measure Gear Tooth Radial Forces	29
8.	Proposed Design for Spring Retainer	34
9.	Proposed Test Gear Shaft and Support Bearing Arrangement for One-Motor Operation	34
10.	Assembly Drawing of Lubricant Tester	44

ACKNOWLEDGEMENTS

The author wishes to recognize the contribution of Professor E. K. Gatcombe. The research was conceived by him and, was his original thoughts and efforts to obtain vital equipment that made the project possible. The author further wishes to express his sincere appreciation to Professor R. W. Prowell for his help and guidance in connection with this project. Others who were most helpful in conducting this research were Mr. Frank A. Glassow, Manager of the Electromechanical Department of the Space Systems Division, Hughes Aircraft Company and Mr. Vance Hoffman of the Vacuum Division of Varian Associates. Mr. R. F. Zahn of the Electrical Engineering Department, Naval Postgraduate School deserves special mention for his work in building the relay circuits and integrating the components that make up the stepper motor power supply. Finally there is my wife, who by her assistance and support made my job so very much easier.

1. Introduction.

This project was to carry on the work started by M. M. Sampsel who designed and built the Lubricant Tester apparatus [12].¹ The research goals set forth when the project commenced were to: (1) provide a power supply for the stepper motors, (2) have gold sputtered on bearing and gear tooth surfaces, (3) install and calibrate a torsion bridge to determine tooth contact forces, and (4) calibrate the installed strain gages to determine radial forces on mating gear teeth. All but the last was accomplished successfully and each in turn is discussed in detail within this report.

While reports commenting favorably on the use of electroplated gold and gold alloy films for vacuum applications are available [1, 2, and 10] there is a dearth of information available on the use of sputtered metallic films for such applications. It is hoped that this project will eventually contribute knowledge to this field.

¹Numbers in brackets designate references in the Bibliography.

2. Developing the Power Supply.

The motive source for the lubricant tester is two stepper motors developed and manufactured by the Space Systems Division of Hughes Aircraft Company, El Segundo, California. The motors provide a high torque output in a low-weight, small sized package of relatively simple construction and with no intermittent electrical contacts. When a pulse is applied to one of the two coils, the armature moves in an axial direction against the action of a helical spring. Ball bearing cam followers on the periphery of the armature act against profiles on the power stroke cam. When the coil is de-energized, the helical spring returns the armature axially until the cam followers encounter the sloped surface of the detent cam. Further motion is prevented by the pre-loaded detent spring that holds the cam followers in the lowest position of the detent cam. This mechanical lock feature was very useful when making static measurements under load. When the cam followers have come to rest, the armature has rotated 40 degrees from its position at the beginning of the pulse. Rotation of the armature shaft is transferred through a three-stage planetary transmission with a step-down ratio of 289:1 to the output shaft. Thus each pulse rotates the output shaft 0.1384 degrees.

The initial requirements for the power supply were that it provide 50 millisecond pulses at 28 volt DC with peak currents up to 30 amperes. [7] It was proposed that the voltage be maintained at a constant value and the current allowed to vary as required. In consultation with Hughes Aircraft Company it was learned that the motors were designed to operate at nearly constant current level. The nominal value being 10.5 amperes per motor with 17.8 amperes as the maximum allowable current through each motor.

The first major problem connected with the power supply was how to obtain the direct current pulses necessary to operate the motors. A signal generator was connected to a small relay that controlled the operation of a larger DC relay which would carry the main power. It was found necessary to add a transistorized relay driver circuit to obtain the necessary power to operate the larger relay reliably, but the project was on its way.

Once it was demonstrated that suitable pulses could be produced and that the pulse repetition rate could be adjusted, some switching method had to be found so that the motors could be reversed. Preliminary investigation of the entire problem had considered generating the pulses by the use of microswitches and a revolving toothed wheel. A modification of this idea with two microswitches was now proposed. While the idea had the merit of simplicity; it also posed the difficulty of proper synchronization with the DC pulses. It was at this juncture that experiments with the Hewlett-Packard Model 5202L Scaler/Timer were begun. While the instruction manual indicated that this equipment might be suitable for the purpose, experimental verification was required. The Scaler/Timer was connected to a signal generator and a dual-beam oscilloscope and the Scaler/Timer's performance observed. The result was eventually successful; proper operation was found to be very dependent on the input signal amplitude. A signal of insufficient strength failed to register with the counter circuit and too strong a signal was counted more than once. The Scaler/Timer required a well-defined square pulse for consistent operation, as any slope on the sides caused the pulse to be counted twice.

Figure 1 is a diagrammatic representation of the power supply system. A. C. line voltage from the wall is taken through a variable autotransformer (variac), a step-down transformer, and a full wave rectifier network. The rectifier output is a smooth DC voltage that can be adjusted over a range of values. During the test, the voltage level was set at a value such that the peak current through both motors was 24 amperes. Initial configurations of the system utilized conventional ammeters, but these were found to give erroneous readings. A calibrated shunt was placed between the rectifier and the pulse relay. A Brush Mark 280 recorder was connected across the shunt and it indicates current by a pen deflection of one amp per line.

A Hewlett-Packard Model 211A Square Wave Generator provides timing pulses for the entire system. The motors were designed to operate at rates up to eleven cycles per second (cps), and Hughes Aircraft Company conducted their Engineering Qualification tests at 2.0 cps. However, the manual dexterity of the author and the convenience of direct conversion from the digital timing clock indicated that 1.0 cps was the best choice. Unfortunately the pulse duration from the Square Wave Generator is far greater than the specifications set by reference [7] of 48 ± 5 milliseconds. To give the correct pulse duration a General Radio Company Type 1217-B Unit Pulse Generator was utilized. Because 1.0 cps is less than the Unit Pulse Generator can supply from its internal circuitry, the Square Wave Generator is used to drive it externally.

A 50 millisecond square pulse is sent from the Unit Pulse Generator to a transistorized relay driver circuit. In effect a small amplifier, it provides sufficient power to reliably operate the control relay. This small control relay has 24 volts DC across it and is used to

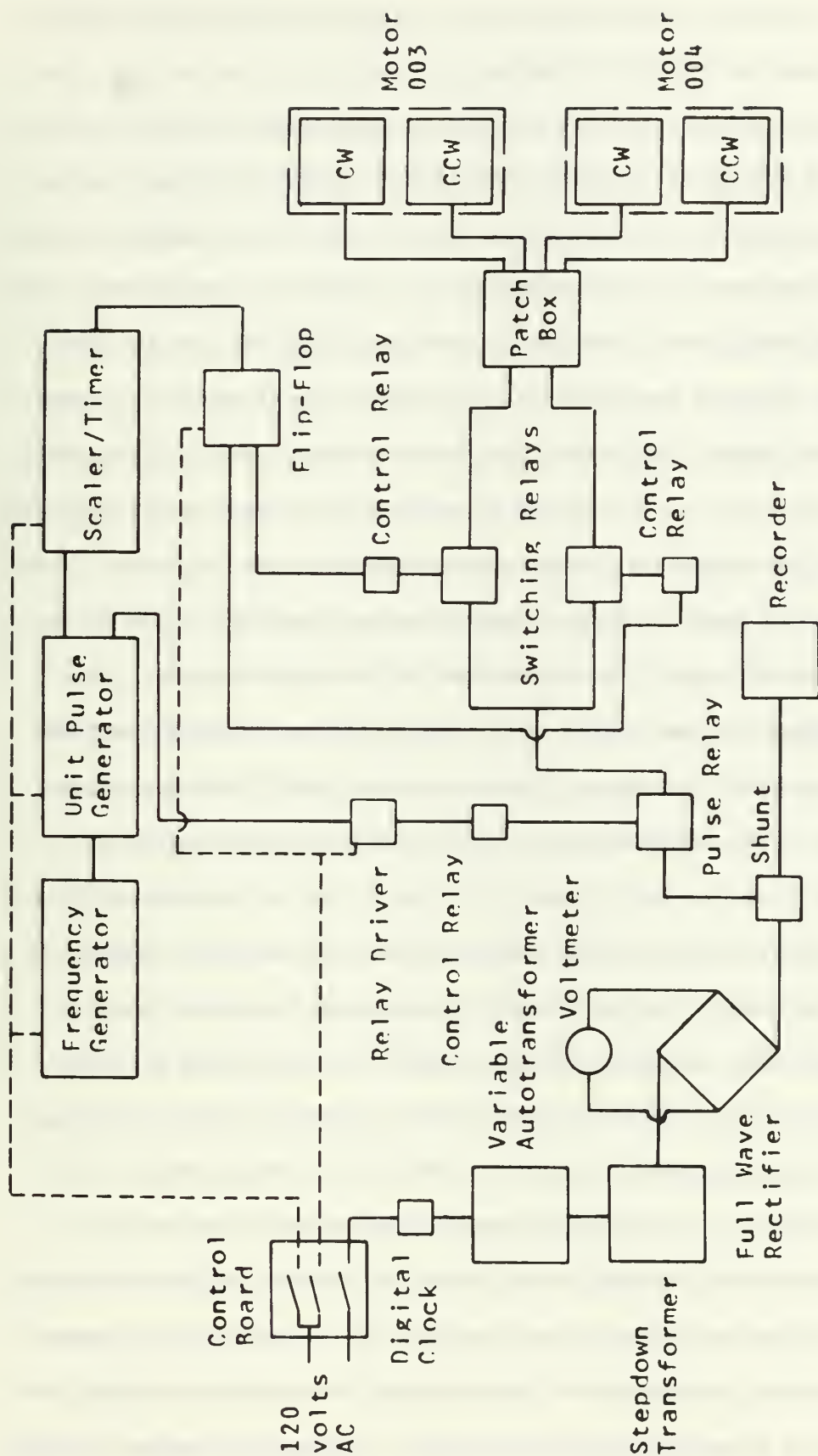


FIGURE 1 - BLOCK DIAGRAM OF THE STEPPER MOTOR POWER SUPPLY SYSTEM

control the operation of the larger power relay that produces 50 millisecond DC pulses to operate the motors.

From a parallel connection, another 50 millisecond pulse goes from the Unit Pulse Generator to the Scaler/Timer. If the pulse and switching relays with their associated auxiliary circuits are considered to be the heart of the power system, then the Scaler/Timer is its brain. With the proper settings on this equipment, the motors can be run in either direction for extended periods or can be made to oscillate over an arc of any desired length. Unlike an electronic counter, the unit can only be set to display the time required to achieve a pre-set number of counts or to display the number of pulses accumulated in a pre-set time. When either of the two conditions mentioned above are reached, the next input pulse triggers a gating pulse and the unit resets itself.

The negative portion of the gate pulse activates a transistorized multivibrator flip-flop circuit. The outputs of the flip-flop circuit are connected to two small control relays similar to that mentioned earlier. Amplification within the flip-flop circuitry provides sufficient power to operate the control relays without the need for a separate relay-driver circuit. The two control relays each operate a larger switching relay that controls the distribution of the pulsed DC power from the pulse relay. Since the two sides of the flip-flop circuit are energized alternately, one or the other of the switching relays is always closed. The gate pulse that signals completion of the pre-set number of counts from the Unit Pulse Generator affects alternate closing and opening of the switching relays and thus the reversal of the direction of rotation by the motors. Indicating lights across the relay contacts show which relay is closed. This is a very useful feature to avoid rotating the motors in the undesired direction. There is a separate

motor coil for each direction of rotation and no provision within the motor to prevent simultaneous energization of both coils. The gate pulse biases the quiescent side of the flip-flop for operation, and momentary operation of this side shuts off the side that had been operating. Thus, for an instant, both switching relays may be closed and power supplied to both directional coils. In consideration of the switching times involved, power through both coils for a short time is not regarded as a problem. Outputs from the switching relays and a common line from the rectifier are taken to the patch box where combinations of individual motor operation can be selected.

The total elapsed steps of the motors, provided that operation is at 1.0 cps, is recorded on an electric digital clock arranged through the main power switch on the control board. The other two switches on the control board are for power to the relay board, transistor power supplies, Scaler/Timer, recorder, and generators. This provides a quick and nearly foolproof way of warming up the system and placing it in a stand-by status.

While the power system just described provides a flexible, easily controlled means of operating the motors, it is still plagued by a few continuing problems. As single components and sub-assemblies were integrated to form the system, the difficulty of feedback arose. Operation of the pulse generation circuits affected the counting circuits within the Scaler/Timer and resulted in erratic operation. Shielded leads and the addition of blocking capacitors largely eliminated this problem. However, it was necessary to take input pulses for the Scaler/Timer from the Unit Pulse Generator rather than from the pulse relay, as might have seemed a better practice. The instability, associated with the gain of

the generators, was mentioned earlier. This problem still persists and the amplitude settings on both the Square Wave and Unit Pulse Generators should be adjusted if the display on the Scaler/Timer indicates a value significantly less than expected or if the Scaler/Timer fails to reset itself. A safety device to prevent the application of continuous DC power to the motors should be incorporated into the design of the system. Hughes suffered the loss of a motor during their testing program when it burned out after continuous power was accidentally applied for a period of about 30 seconds. The author never left the very immediate vicinity when main power was energized, but this restriction would be inconvenient or impossible during actual failure tests. The entire system is adversely affected by fluctuations in the line voltage. When operating the motors, other electrical equipment should not be started or secured. The fluctuations cause the pulse relay to give two pulses to the motors in rapid succession. While this double pulse will occasionally occur without apparent reason, it is believed to be caused by a fluctuation that the operator didn't notice. The operation of a single small light near the power supply produced double pulsing. While double pulsing is admittedly disconcerting, it appears to have no adverse effect on the motors if they are being operated at a low pulse repetition rate. However, the electric clock doesn't detect the double pulse, while the Scaler/Timer does take account of it.

3. Sputtering Processes.

Sputtering is a material deposition technique in which positive ions are given high kinetic energies and directed toward a sample of the material to be deposited (the target). A transfer of momentum process occurs between the ions and the atoms of the target. If sufficient energy is transferred to overcome the binding energy of the lattice, atoms of the target structure are knocked out or sputtered from their positions in the target lattice. The sputtered atoms leave in a random manner and many will be intercepted by the recipient surface if it is placed sufficiently close to the target. Re-combination with other sputtered atoms occurs and a thin film is produced on the recipient surface (the substrate).

Many variations of the basic process are possible [3 and 6]. Reactive substances may be introduced so that re-combination on the substrate yields a material different from the target. A similar effect may be obtained by using multiple targets or a stream of reactive ions to affect the sputtering. Sequential sputtering with targets of different materials will produce consecutive layers of these materials.

Two distinctly different techniques are utilized to provide the positive ions for the sputtering process. The first, known as DC or cathodic sputtering, produces ions by creating a glow discharge between a negatively charged target (the cathode) and the substrate. This technique is only possible with conducting materials [3]. The alternate technique allows the use of nearly any target material, whether conductor or insulator. This procedure, known as rf sputtering, creates an ion plasma by application of a high frequency alternating potential. The difference in mobility between the ions and electrons in the alternating field neutralizes the positive charge which

accumulates on a dielectric surface when it is bombarded by ions. Frequencies near 13.56 megacycles are most generally used [9].

A further refinement of the above techniques is called substrate biasing. Here the substrate is biased slightly negative with respect to the ion field. The result is that some ions are drawn directly to the substrate and sputter off a small portion of the coating that is being applied from the target. The resulting film is improved in density and adhesion [8]. A fairly uniform film can be applied over a surface even when portions of that surface are out of direct line-of-sight with the target [6]. This last would be especially applicable to the sputtering of gear teeth.

The actual sputtering for this project was done by Varian Associates, Palo Alto, California, on an RF Plasma Coil Sputtering Apparatus manufactured by that company. A vacuum chamber that contained the objects to be sputtered was first evacuated and then back-filled with argon gas to a pressure of about 1 micron (1×10^{-3} millimeters of mercury). To produce ions from the argon gas, 1.2 kilowatts of power at 13 megacycles was then supplied to a nine inch diameter, two-turn coil. A negative potential of 1000 volts DC was then applied to the target, a copper plate electroplated with gold, positioned about one inch from one end of the coil. Argon ions with energies of about 1000 electron volts impact against the gold target and exchange momentum with the gold atoms. The gold atoms ejected from the lattice structure possess considerable energy and when they collide with the substrate they are driven into the surface lattice structure. The effect of the large arrival energy is adjudged responsible for the increased adhesion shown over electroplated or vapor deposited films.

In general a sputtered film can be expected to be tightly adhesive and of nearly the same density as the bulk target material. However, there is no certainty that the sputtered film will have the same structure or properties as the target or the bulk target material. Sputtered gold films are usually polycrystalline with very small grain sizes and in the tightest packing configuration the face-centered-cubic structure of gold will allow [14].

Varian Associates applied a film 50×10^{-6} inches thick to all surfaces to be sputtered except the gear teeth. Because it was desired that failure occur first on the teeth rather than on any of the shaft bearing surfaces, the film thickness on the teeth is 17×10^{-6} inches. With one exception the actual film has been quite adherent through calibration and test operations. When the torsional strain gages were being applied the journal surface of the test gear shaft was wrapped with Teflon backed adhesive tape to protect it. When the tape was removed, a small section of the film was pulled up. Varian offered to re-sputter the surface free of charge, but the time required to properly prepare the surface by grinding and polishing was considered excessive. Calibration experiments have removed further portions of the film adjacent to the original break, but removal to the steel surface has not occurred elsewhere. A number of factors influence adhesion of sputtered films, and peeling will occasionally occur even under the most favorable conditions [3].

4. Calibration of Strain Gages to Measure Contact Forces.

The technique proposed by Sampsel [12] for determining tooth contact forces from a properly calibrated torsion bridge was tested as soon as Varian Associates had completed the sputtering. It was decided to mount a four gage torsion bridge as illustrated in reference [11] but with all the gages on one end of the test gear shaft. This configuration was chosen to simplify installation and lead wire problems. By the use of short connecting wires between sides of the shaft, only four lead wires needed to be taken to the strain indicator. An attractive bonus of this arrangement was that it reduced the possibility of damage to the gages when the test gear was assembled on the shaft. The gages actually installed were two Baldwin-Lima-Hamilton type FAED-06-12S6 gages. These are two-element foil rosettes on an epoxy backing with the elements oriented 45° to the gage axis. Further details on the gages and the application techniques used are found in an appendix.

Application of the first four gages was not satisfactory. Baldwin-Lima-Hamilton (BLH) EPY 400 cement was used for this application. This is a two-part liquid and powder cement that is mixed together in pre-measured quantities before use. Despite efforts to thoroughly mix the constituents, small lumps of the powder were distributed throughout the resulting mixture. When the gages were examined after the heat cure they presented a speckled appearance. There seemed to be numerous tiny lumps or air pockets under the epoxy backing. On the basis of this appearance, partial bonding of the gage to the underlying surface was presumed. The cement used was three months out of date when mixed, and this may have contributed to the poor results. Measurements with a Vacuum Tube Voltmeter indicated that elements of two gages had been

damaged during installation so as to render the gages useless. Another gage was ruined when the foil broke while soldering the connecting wires. Only one two-element gage remained and no spares from the same lot were available at the school. At the recommendation of the BLH representative, a new type of gage cement was obtained along with replacement FAED-06-12S6 gages. EPY 600 cement is a two-part liquid-liquid cement that gives a very thin glue line around the gages and promotes good bonding on curved surfaces as any entrapped air pockets are easily smoothed out. Two gages applied with EPY 600 showed no visual indications of partial bonding and have performed most satisfactorily in tests since the time of their application.

As soon as installation of a torsion bridge was successfully completed, calibration of the bridge was begun. The test gear, springs, and spring retainers were assembled on the test gear shaft and the combination clamped between the shaft pillow blocks. Tooth profiles on the calibration tool were matched with corresponding profiles on the test gear. Weights applied to the end of the calibration tool exerted a torque on the shaft-spring combination that unbalanced the torsion bridge. The bridge unbalance was converted directly into units of strain by a BLH Type N Portable Strain Indicator. The mounting plate was clamped to the workbench to restrain its rotation, and the rotation angle of the calibration tool with load (\emptyset) was determined with a machinist's protractor level. The effect of the calibration tool's rotation on the lever arm for determining torque (R1) was accounted for by the formula:

$$R1 = 12.0 \cos \emptyset + 2.0 \sin \emptyset$$

Here 12.0 is the length in inches of the straight portion of the calibration tool and 2.0 is the radius of the test gear pitch circle in inches. See Figure 2 for further illustration.

Typical results of these tests are shown in Figure 3. Each test showed a separate and distinct path for the application and removal of the load that produced a double valued curve for torsional strain as a function of applied torque. A straight line single-valued curve was obtained in one test, but the conditions of the test were such that sufficient force was applied to the end of the calibration tool to bend it.

Failure to obtain the expected results brought careful re-evaluation of the test procedure. After examination of the apparatus three alterations were made. The use of a single calibration tool introduced unbalanced forces normal to the bearing surface of the shaft. Friction forces could then oppose the applied force that produced the torque. To balance these forces and give an operation such as Sampsel intended, the use of two calibration tools was proposed. The Machine Facility speedily fabricated the desired items. They used somewhat heavier stock and left more material at critical points so as to provide greater resistance to bending.

Blueprint specifications for bearing surface clearances were checked against the table given by Faired [4]; the clearances specified were tighter than those given for RC 1 close-sliding fit. The tight fits were further confirmed by the difficulties encountered when assembling or disassembling the apparatus; many pieces fitted together only one way and rather snugly at that. With fine emery paper and crocus cloth the clearances on the bearing surfaces were eased by hand to the specifications for an RC 2 sliding fit. The values obtained after completion of

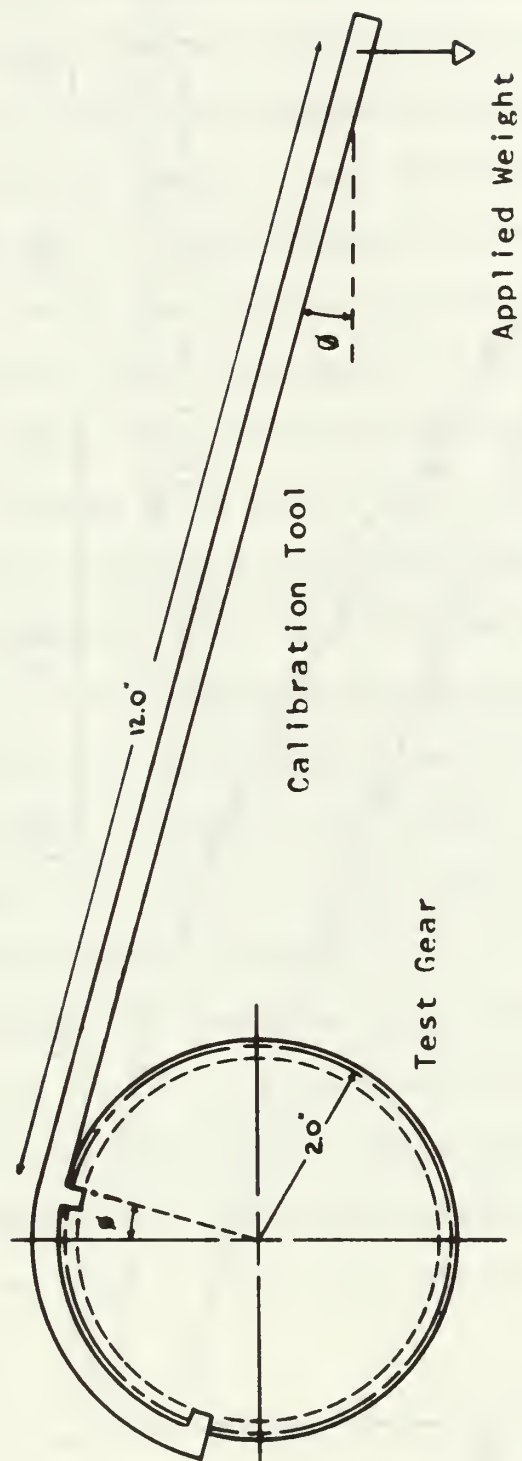
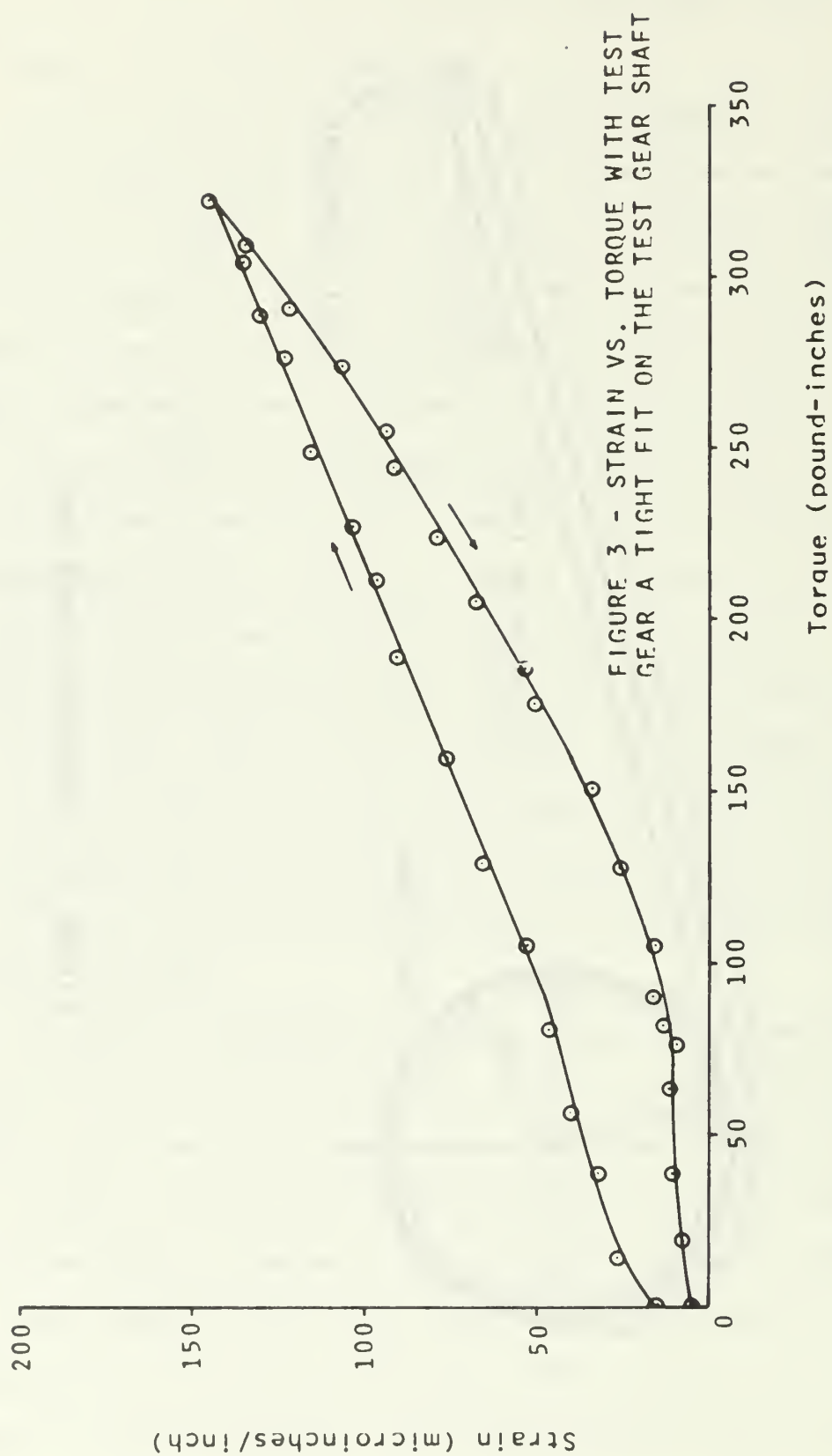


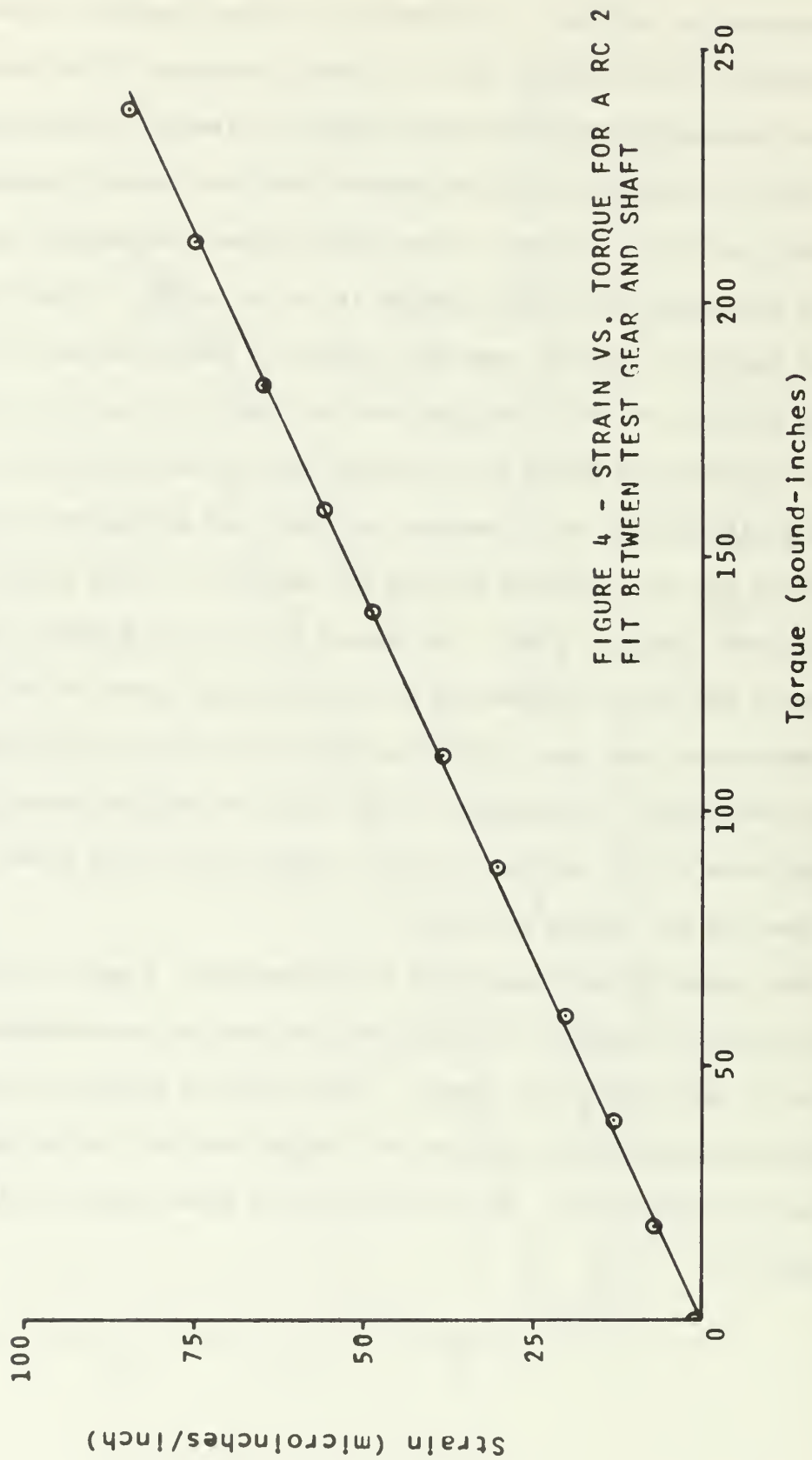
FIGURE 2 - TORSIONAL LEVER ARM



this work are as follows: (1) diameter of journal surface of the test gear measured 1.1259 inches, and (2) journal surfaces of the drive pinion supports measured 0.8910 and 0.8914 inches in diameter respectively.

When the apparatus was disassembled after the initial attempts to calibrate the torsion bridge, it was noticed that the springs had rubbed against the shaft, noticeably gouging it at one point. It was earlier noticed that when the test gear was rotated, a couple was set up that twisted the ends of the spring against the shaft. It was not realized at the time that the force of the spring on the shaft could be so large. Teflon split bushings to fit between the shaft and spring were proposed. Because of the time delay in getting the material, it was decided to use Teflon backed adhesive tape. Five layers of one inch masking tape were applied to the shaft, followed by two overlapping layers of Teflon tape. This was done at each end of both springs where they tended to rub against the shaft. In addition, Teflon tape was wrapped around the straight ends of the springs to give a tighter fit in the holes in the test gear and the spring retainers.

The result of the changes was quite apparent. Figure 4 is the curve for strain as a function of torque that was used in the attempted calibration of the tooth strain gages. This figure is typical of the others obtained when the Teflon tape was not damaged and the torsion gages were positioned horizontally. The set-up used for these tests is illustrated in Figure 5.



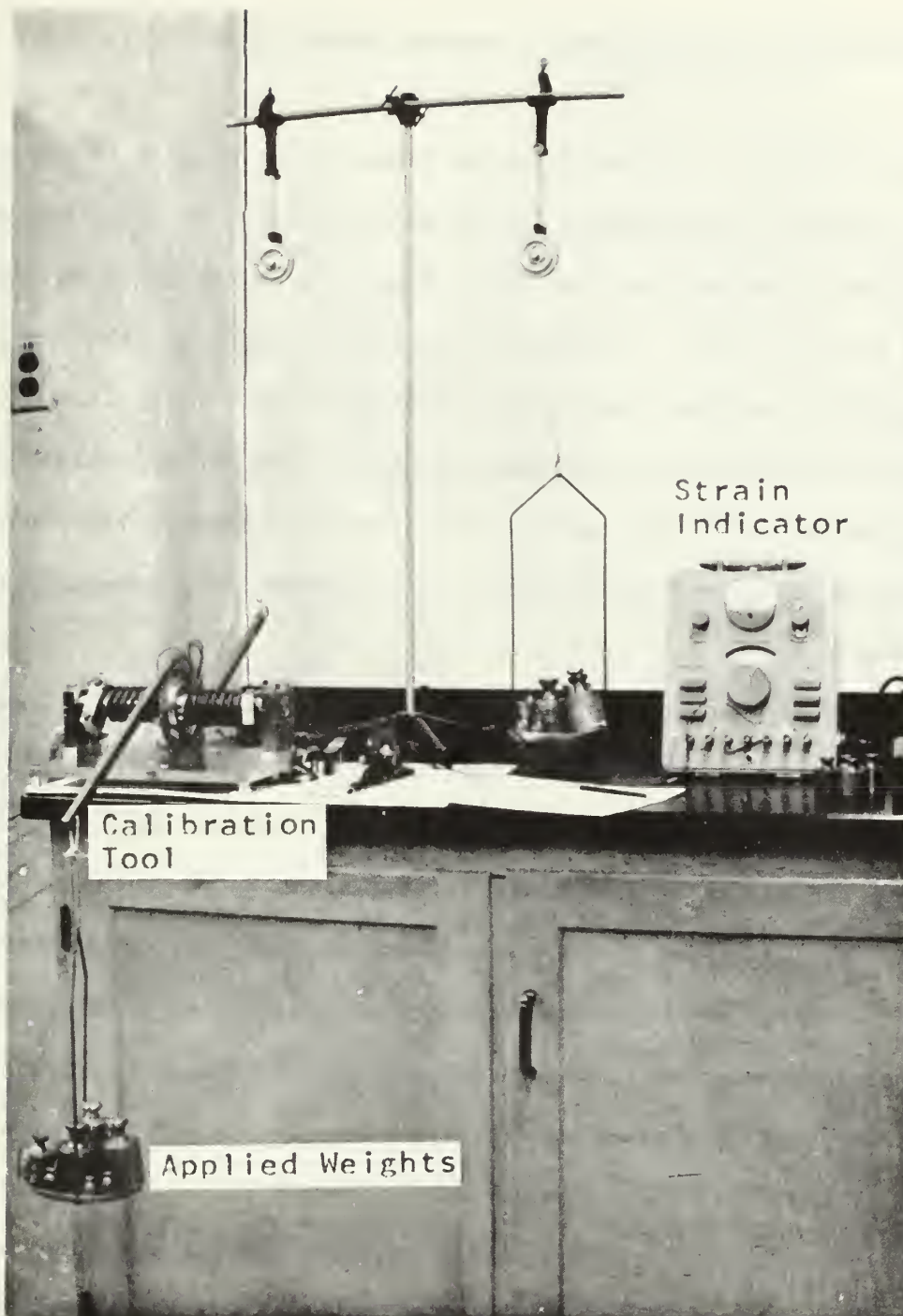


FIGURE 5 - TEST SET-UP FOR CALIBRATION OF THE TORSIONAL STRAIN BRIDGE

5. Calibration of Strain Gages to Measure Radial Force on the Gear Teeth.

Sampsel proposed that the friction force acting tangent to the point of tooth contact be considered a purely radial force. He justified this approximation by saying that the radial component of the friction force is 94% of its value [12]. To measure the radial forces on the teeth, a four-gage bridge was installed on the test gear. This bridge had an active gage on each side of the teeth in contact and two compensating gages mounted on teeth well removed from those in contact. This configuration was to give a bridge factor of 2.0 and be sensitive only to radial forces.

As a part of his study of dynamic loads on spur gear teeth, Umberger determined the static loads on two gear teeth in contact [13]. Figure 6 is a representation of the results he obtained for the variation of strain with the component of contact force tangent to the pitch circle and contact radius. Note how the same value of force will give different strain readings depending on the point of application to the tooth. Umberger neglected radial forces on the gear teeth because he found them to be of small magnitude and their measurement interfered with the operation of his other strain measuring instruments.

Umberger measured forces on gear teeth where the point of contact was well lubricated by an oil film. Under these conditions of viscous lubrication any friction force could be neglected with respect to the radial component of the contact force. However, for solid film lubrication, the frictional force may be of the same order of magnitude as the radial component of the contact force. It was anticipated that the combination of the larger radial force, and the amplification due

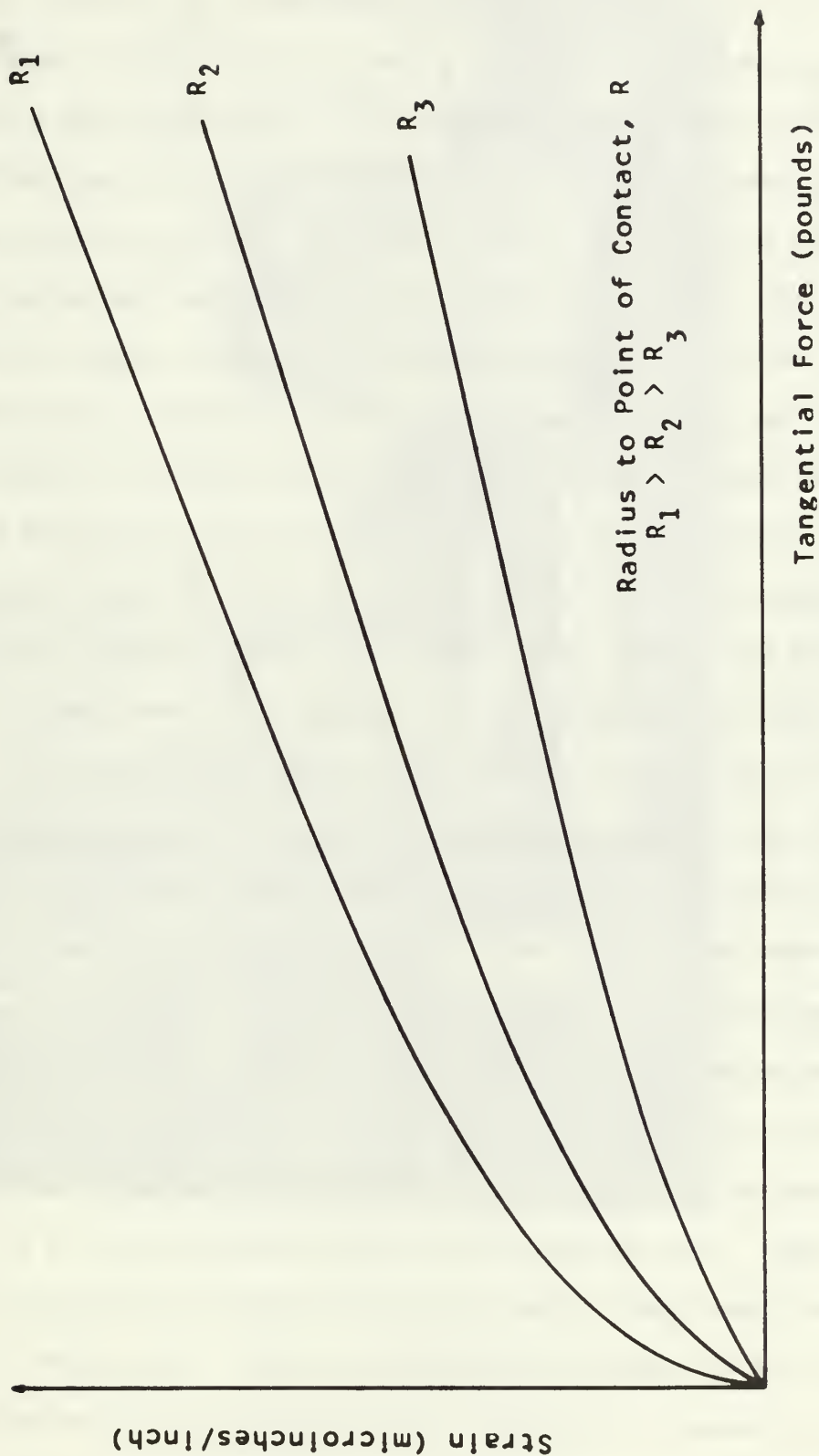


FIGURE 6 - REPRESENTATION OF STRAIN AS A FUNCTION OF THE COMPONENT OF CONTACT FORCE TANGENT TO THE PITCH CIRCLE AND POINT OF CONTACT

to the bridge factor of 2.0, would allow data to be collected. A complication is that more than one tooth is in contact at the same time since the Lubricant Tester gears have a contact ratio of 1.69. Umberger avoided the problem of multiple contact points by using a gear with a single tooth that was mated with the instrumented teeth. He could vary the load on the teeth in contact while they were locked in place, but this can not be done with the Lubricant Tester. To equalize the load between both springs on the Tester, it is necessary to generate contact forces by rotating the gears with the motors. While the radial forces between just two teeth in contact could not be established for the Lubricant Tester, it was hoped that a relationship could be established for radial strain as a function of contact force and point of contact, and that a different but similar relationship existed for each type of lubricating film. Thus a comparison of strain reading for a known contact force and point of contact would be indicative of film's performance.

The strain gages mounted on the test gear are in an exposed position; this is particularly true during assembly and disassembly of the apparatus. By the time calibration tests were started, only two of the original six gages were undamaged. These two were connected to form a two-gage bridge with one active gage on a tooth in contact. The set-up for these tests is shown in Figure 7. BLH Type N Portable Strain Indicators were used to measure strains from the torsion bridge and the bridge on the test gear teeth. The instrumented teeth were backlighted with a 12 volt grain-of-wheat bulb so that the point of contact appeared as a dark spot on the line of light that surrounded the teeth. The distance from the point of contact to the root of the instrumented teeth was determined by the graduated, traversing cross-hair of a small telescope. The cross-



FIGURE 7 - TEST SET-UP FOR CALIBRATION OF STRAIN GAGES
TO MEASURE GEAR TOOTH RADIAL FORCES

hair was calibrated by determining the apparent distance between two lines 0.25 inches apart on the test gear.

There was a difference of about 13 ohms between the two strain gages installed on the teeth for no apparent reason. The resistance to ground readings were nearly the same for both and the lead wires showed negligible resistance. To compensate for this resistance difference, a decade resistance bank was added to one arm of the bridge. Sufficient replacement gages were not available and it was decided to continue with the tests due to the short period of time that remained.

When calibration tests were actually performed, some change in radial force was detected as the point of contact moved along the side of the instrumented teeth and the contact force varied. However, the data for different runs could not be compared as a true unstressed strain reading or "zero" could not be established. Readings taken when the apparatus was disassembled before and after a test run varied greatly. In some cases this variation was about half of the greatest difference in readings obtained during the test. On this rock the whole series of calibration tests foundered. Because strain readings for any one test could not be compared to a "zero" value it was not possible to determine any relation for the radial force. However, it should be stated that the trend of readings was the same for nearly all the tests. This indicates that the desired relation may in fact exist.

6. Suggestions for Re-design.

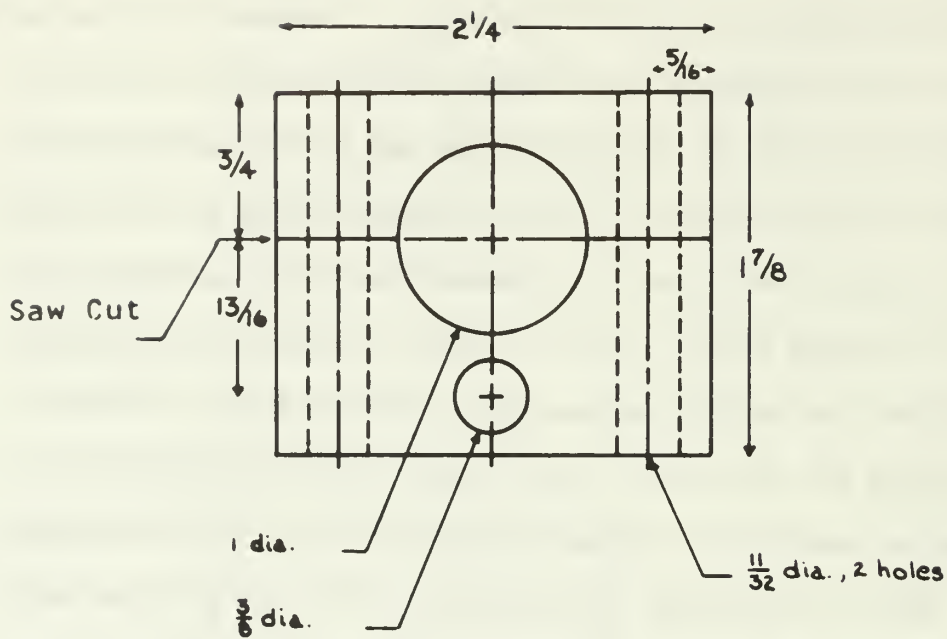
While no design changes were anticipated when the project was undertaken, the experience gained during testing indicated the Lubricant Tester would benefit from some alterations. Those changes fall into two broad categories: (1) changes for the convenience of the researcher and/or the purpose of having the Tester operate more nearly as originally envisioned, and (2) changes that affect the fundamental assumptions that Sampsel made when he designed the Lubricant Tester.

Foremost among the first category is the problem of proper positioning and alignment when the Tester is assembled for each test. The specifications that call for a $3/8$ inch stud to pass through a $7/16$ inch hole allows too much clearance. This prevents accurately repositioning the part after dis-assembly and introduces misalignment between motor and test gear shafts and unwanted bending stress. To alleviate this problem, guide pins should be fitted between mating pieces or the holes bushed to ensure a tighter fit by the stud. The present motor supports are not satisfactory. They should be replaced with a cradle assembly that supports the motor body and journal surface of the driving pinion as a complete unit. This cradle should allow easy removal of the motor and driving pinion without disturbing the test gear shaft. It would be desirable to lengthen the test gear shaft so as to increase the distance between the spring retainers and the shaft pillow blocks. The distance presently available is just sufficient to mount the strain gages for the torsion bridge. Because of the distances involved, the possibility of damaging these gages during assembly is great. In addition a larger size strain gage could be mounted on the longer shaft. The FAED-12-12S6 gages are a stock item at the West Coast office of Baldwin-Lima-Hamilton, while

FAED-06-12S6 gages have to be special ordered from the factory, often with considerable delay. The positions that the present spring retainer blocks can assume are limited because of interference with the adjacent motor housing. This is an inconvenience as it limits the positions that the test gear can assume; spring retainers such as these shown in Figure 8 would eliminate this interference. The Teflon split-bushing mentioned in section four should be designed so that the spring will be prevented from rubbing on the test gear shaft. If rolling contact bearings suitable for vacuum operation and with a sufficiently long life expectancy can be obtained at a reasonable cost, they should be installed to replace the present journal surfaces. After calibration tests and operation of the motors for only about 20,000 steps the sputtered film has been removed on a small section of the horizontal surfaces while the remainder of the sputtered film shows no indication of peeling. However, if peeling did occur over a large area during an actual life test, it could introduce friction forces that had not been present during calibration. A bearing surface which has lost a portion of its sputtered film must be ground and polished prior to re-sputtering. This machining would change the fit between bearing and journal, and may soon lead to excessive play in the bearing. A rolling contact bearing could provide a smooth rolling action and allow replacement without disturbing the bearing fit. Any such bearing must also be suitable for use at the temperature extremes that will eventually be encountered during life tests.

A fundamental assumption made by Sampsel, that the two contact areas on opposite sides of the test gear will share the load equally, may be false. When operating with both motors, there exists a separate driving force on each side of the test gear. Equal load sharing under these

conditions would seem to depend on having the motion of the driving pinion for each current pulse the same for both motors. Because of slight variations of the motor mechanisms, the planetary transmissions, and the external gear teeth, the angle of rotation may not be the same for each pulse. Various configurations of contact between mating teeth have been observed. On several occasions, it appeared that there was contact on both sides of the same tooth. On one occasion the teeth were observed to rotate without any contact between them, indicating that the other gear was carrying the entire load. In light of these circumstances it may be deemed advisable to re-design the Lubricant Tester for one motor operation. The use of bearings placed close to the test gear and two shafts coupled by the spring may sufficiently reduce the effects of bending caused by the gear separating forces. Figure 9 illustrates one possible configuration where the helical spring transmits torque between two shafts.



all dimensions in inches $\pm 1/64$, break all sharp corners, material- $\frac{5}{8}$ stainless steel

FIGURE 8 - PROPOSED DESIGN OF SPRING RETAINER

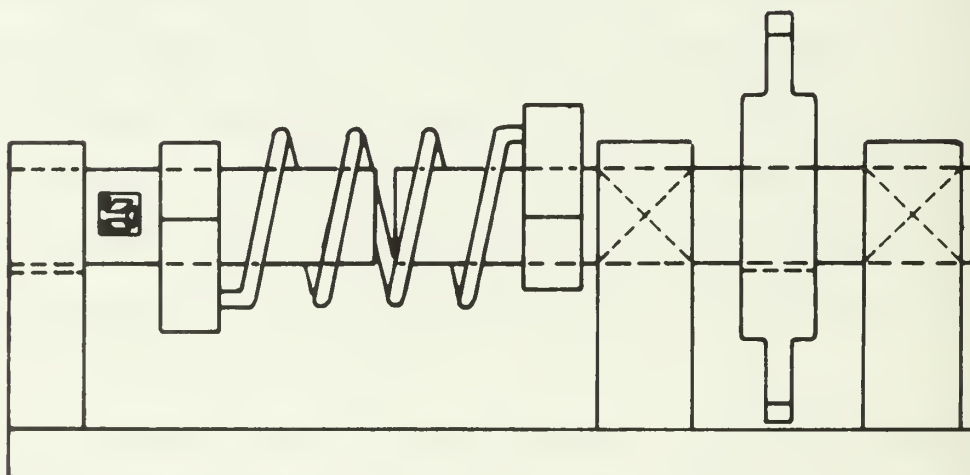


FIGURE 9 - PROPOSED TEST GEAR SHAFT AND SUPPORT BEARING ARRANGEMENT FOR ONE-MOTOR OPERATION

7. Conclusions and Recommendations.

While not all the research goals undertaken have been successfully completed, the objectives of the Lubricant Tester program are closer to realization. The stepper motors have a power supply system that is flexible enough to meet the needs of future researchers and the contact forces can be determined from strain readings on the torsion bridge. If the determination of film failure through measurements of the radial force exerted on mating teeth is not possible, determination of failure should be possible by other methods. Perhaps an optical "light pipe" could be used to visually examine the film for evidences of failure. The practical experience with the apparatus gained during this project has brought forth suggestions for its improvement.

It is strongly recommended that some provision for improved alignment and accurate repositioning of component parts after disassembly be incorporated into the design before further work is undertaken with it. In addition, experience has shown that the use of Teflon tape to protect the test gear shaft from contact with the springs is at best a substitute for Teflon bushings. Before extensive use of the motors is undertaken, some form of safety device to prevent the application of continuous power to the motors must be installed. The addition of an isolation transformer or a line filter may isolate the power supply system from the voltage fluctuations that cause "double pulsing." Investigation of this possibility is recommended.

A careful record has been maintained of current pulses applied to each motor. The interior mechanism of the motors and planetary transmissions are coated with a dry-film lubricant. The motors can be disassembled and this coating renewed. However, it is most important that

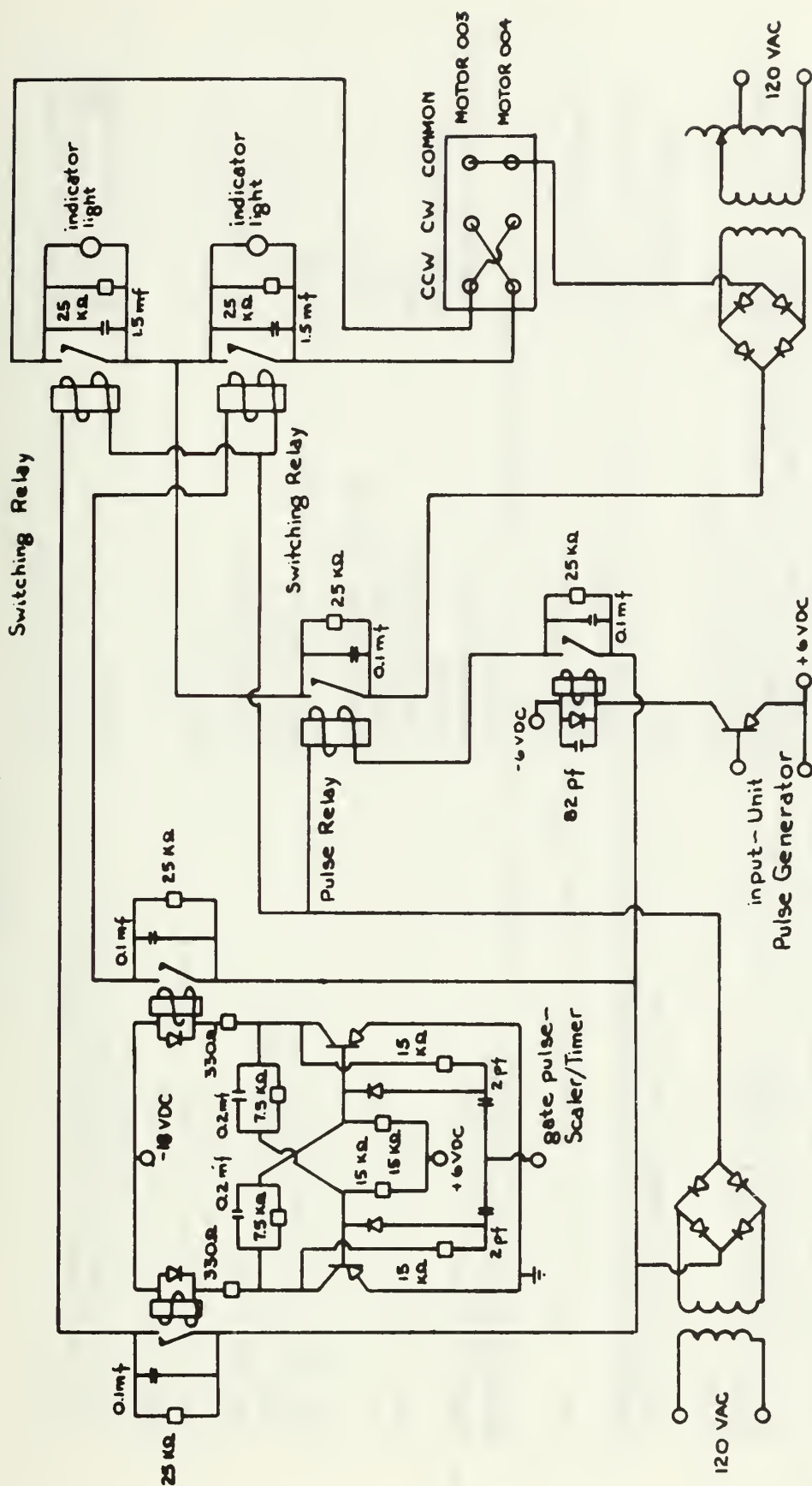
no wear on the metal of internal parts occur as the possibility of obtaining replacement spare parts is considered very slight. While Hughes Aircraft Company obtained lifetimes in excess of 200,000 pulses [7], post-test disassembly indicated some metal wear in all cases. For this reason a more conservative figure of perhaps 100,000 pulses should not be exceeded to avoid the possibility of metal wear.

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APPENDIX I



Wiring Diagram of Pulse and Switching Relay Circuits

APPENDIX II

Stepper Motor Power Supply Equipment and Settings

Equipment	Control Function	Setting
Hewlett-Packard Square Wave Generator Model 211A, Serial 05582	Range Frequency Amplitude (600 ohm)	1X 1.0 3
General Radio Company Unit Pulse Generator Type 1217-C, Serial 1272	PRF Δ F Pulse Duration Range Amplitude	external drive low 5.0 10-100 millisec. 60
Hewlett-Packard Scaler/Timer Model 5202L, Serial 013469	Polarity Display-Preset Preset Sampling Mode Storage (back) Signal Input (back)	+ time(sec) - count steps desired auto off normal
(3) Hewlett-Packard Power Supply Model 721A, Serial 5H5040, 2385, and 024-07836	Short Circuit Current Meter Range Voltage Adjust	50 ma 10 or 30 VDC as required
Precision Scientific Company Digital Electric Clock Type 69230	On-Off	on
Superior Electric Company Variable Autotransformer Type S948, Serial 257	Output	as necessary to obtain 24 amperes through both motors

Equipment	Control Function	Setting
American Transformer Company Step-down Transformer Type OF, Serial 263239	Step-down Ratio	220/110
Brush Two-Channel Recorder Mark 280, Serial 013434	Sensitivity Chart Speed	0.5 millivolts/line as desired
Westinghouse DC Voltmeter Type PX-4, Serial 2357110	Full Scale Deflection	50 VDC
General Electric Calibrated Shunt, Serial NCP 69E377		

Other Test Equipment

(2) Baldwin-Lima-Hamilton Portable Strain Indicator Type N, Serial 004624 and 004625

General Radio Company Decade Resistor Type 1432-M, Serial 22085

Telescope with Traversing Calibrated Eyepiece (Wm. Gaertner & Co.) obtained from Physics Department, Naval Postgraduate School.

Strain Gage Data

The torsion bridge is composed of two Baldwin-Lima-Hamilton FAED-06-12S6 gages from lot 262. These gages are two-element foil rosettes with each element inclined 45 degrees from the gage axis. The gage factor is $1.96 \pm 2\%$ and the resistance of each element is 120.0 ± 0.2 ohms. The gages were applied with EPY-600 cement which is also manufactured by Baldwin-Lima-Hamilton. This is a two-part, liquid-liquid cement that can be heat cured to withstand high temperatures. After installation the gages were cured at 310 degrees F for three hours; this treatment should render the gages effective over the temperature range-250 to 400 degrees F.

The strain gages on the test gear teeth were installed by Sampsel. Details of the gages and their installation are contained in reference [12].

APPENDIX III

Assembly Directions

1. Select the desired direction of gear rotation. For a given contact force, determine the number of teeth on the test gear that will pass through contact while winding the springs to achieve this selected load. For future reference this number of teeth shall be called "off-set."
2. Assemble stepper motor serial number 004 to the fixed motor support. Select and mark the tooth on the driving pinion that will contact the instrumented tooth on the test gear after the two gears have been rotated to generate the contact force. Fasten the support and motor to the mounting plate but do not tighten the nuts on the mounting studs. Slip the driving pinion on the splined section of the motor output shaft. Select a combination of the mating surfaces that most closely gives the same off-set as selected for the test gear. If the correct value of off-set cannot be obtained with the splines, rotate the motor with the power supply. Fit a drive gear support to the journal surface of the pinion but do not fasten it securely to the mounting plate.
3. Put the test gear and both springs on the test gear shaft, exercising proper care to avoid damaging the torsion strain gages. Connect the torsion bridge to a portable strain indicator and determine an unstressed strain reading. Support the shaft on the shaft pillow blocks with the axis of the torsion strain gages horizontal. Do not disconnect the torsion bridge. Adjust the shaft and driving pinion in an axial direction so that contact occurs near the midpoint of the face

width on both gears. The shaft can only be adjusted a limited amount because of the torsion strain gages. Carefully tighten the caps on the pillow blocks while observing the strain indicator. The nuts must be sufficiently tight so that the cap prevents rotation of the shaft but uneven tightening will twist the cap. This undesired twisting will introduce bending in a horizontal direction that the design sought to avoid. It was found this could be avoided if the nuts were selectively tightened until the strain reading was the same as the unstressed reading.

4. Check the alignment between the axis of the test gear shaft and the motor axis. When these axes are parallel, tighten the supports down firmly. Fit the spring retainers on the shaft but do not tighten. Twist the end of the spring in the retainer in the opposite direction to that of test gear rotation and clamp the retainer securely to the shaft.

5. Select and mark the tooth that is to mate with the instrumented tooth on the other driving pinion. Slip the pinion on the output shaft of stepper motor serial number 003 and fit the other drive gear support on the bearing surface of the pinion. After allowance for off-set in the direction of rotation, place this subassembly in the adjustable motor support. Insure that contact is near the midpoint of the face width and set up the nuts on the motor only finger tight. Adjust the position of the drive gear support to match the holes in the mounting plate and secure the nuts finger tight. Twist the motor collar in the direction of test gear rotation and carefully tighten the nuts. Again observe the strain indicator and adjust the nuts so the reading is the same as for the unstressed condition.

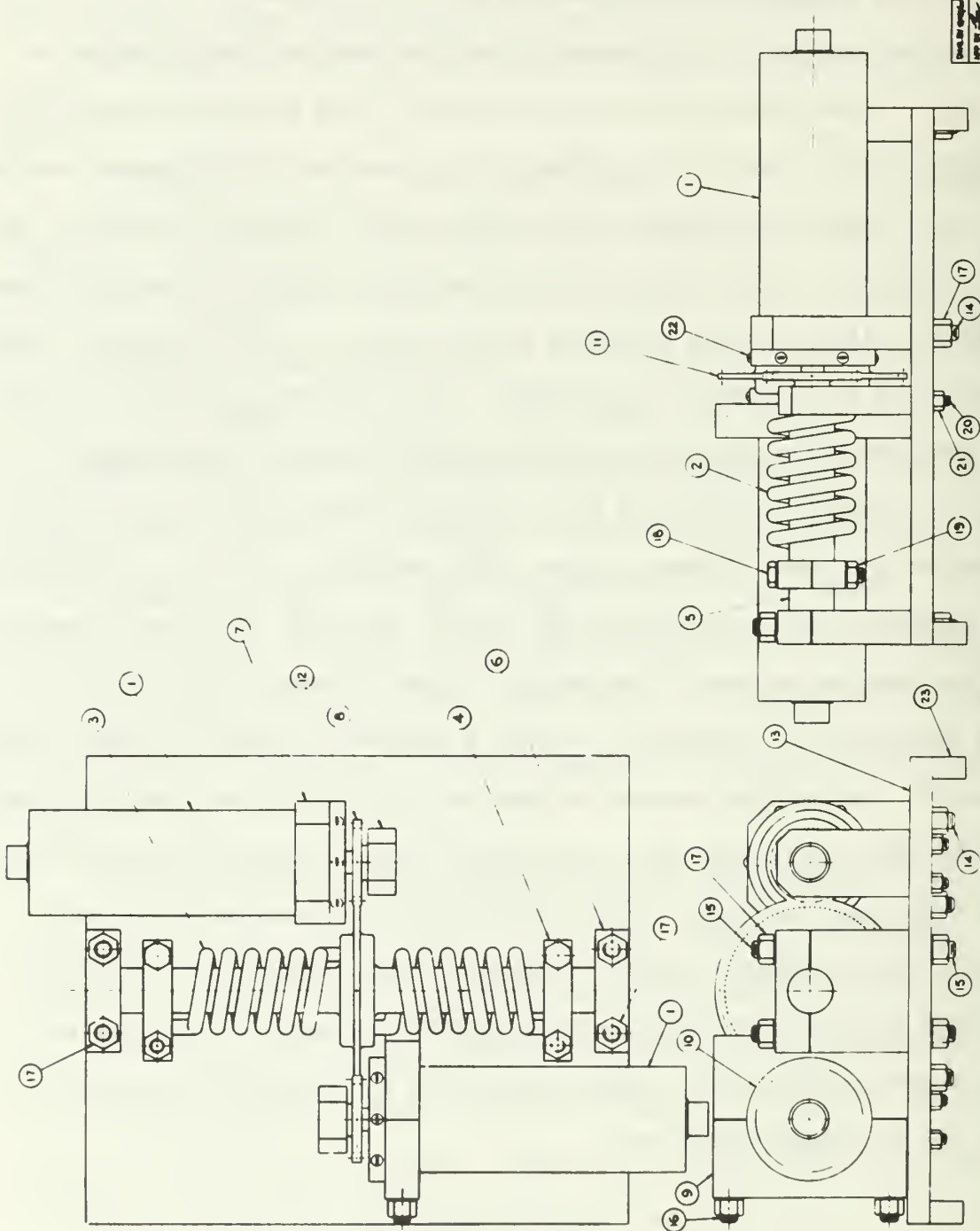


FIGURE 10 - ASSEMBLY DRAWING OF LUBRICANT TESTER

NO.	DESCRIPTION	QTY
1	STEEL MOTOR	1
2	SPRING - LH	1
3	SPRING - RH	1
4	TEST GEAR SHAFT	1
5	TEST GEAR BLOCKS	2
6	DRIVE GEAR SUPPORT	1
7	MOTOR SUPPORT	1
8	DRIVE GEAR SUPPORT, ADJUSTABLE	1
9	MOTOR COLLAR	1
10	TEST GEAR	1
11	DRIVING PISTON	1
12	TEST GEAR	1
13	DRIVING PISTON	1
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100	DRIVING PISTON	1

SCALE 1/2" = 1" 28 MARCH 1968 5-504

APPENDIX IV

Error Analysis

<u>Quantity</u>	<u>Symbol</u>	<u>Uncertainty</u>
Applied weight	W (pounds)	0.05%
Length of calibration tool	L_t (12 inches)	1/64 inch
Radius of test gear base circle	R_b (1.88 inches)	0.0002 inch
Radius of test gear pitch circle	R_p (2.00 inches)	0.0002 inch
Angle of rotation	\emptyset (degrees)	0.1 degrees
Strain reading	e (microinch/inch)	3 microinch/inch

1. Torsional Lever Arm, R_l

$$R_l = L_t \cos \emptyset + R_p \sin \emptyset$$

$$\delta R_l = \left[\left(\frac{\partial R_l}{\partial L_t} \delta L_t \right)^2 + \left(\frac{\partial R_l}{\partial R_p} \delta R_p \right)^2 + \left(\frac{\partial R_l}{\partial \emptyset} \delta \emptyset \right)^2 \right]^{1/2}$$

$$= \left[(\cos \emptyset \delta L_t)^2 + (\sin \emptyset \delta R_p)^2 + (-L_t \sin \emptyset \delta \emptyset + R_p \cos \emptyset \delta \emptyset)^2 \right]^{1/2}$$

2. Applied Torque, T

$$T = WR_l$$

$$\delta T = \left[\left(\frac{\partial T}{\partial W} \delta W \right)^2 + \left(\frac{\partial T}{\partial R_l} \delta R_l \right)^2 \right]^{1/2} = \left[(R_l \delta W)^2 + (W \delta R_l)^2 \right]^{1/2}$$

3. Contact Force, F

$$F = \frac{T}{R_b}$$

$$\delta F = \left[\left(\frac{\partial F}{\partial T} \delta T \right)^2 + \left(\frac{\partial F}{\partial R_b} \delta R_b \right)^2 \right]^{1/2} = \left[\left(\frac{\delta T}{R_b} \right)^2 + \left(-\frac{T \delta R_b}{R_b^2} \right)^2 \right]^{1/2}$$

Table of Uncertainties for Some Individual Points

<u>ϕ</u>	<u>T</u>	<u>δT</u>	<u>$\delta T(\%)$</u>	<u>δF</u>	<u>$\delta F(\%)$</u>	<u>δe</u>	<u>$\delta e(\%)$</u>
5.1	35.54	0.363	1.04	0.193	1.02	3	25.00
13.6	115.88	0.164	0.14	0.087	0.14	3	7.50
24.3	241.90	0.409	0.17	0.218	0.17	3	3.50

The numerical result of individual points indicates that the uncertainty in these points is due principally to the uncertainty in strain measurement. The maximum deviation of individual data points from a best fit line is δe of 2.0 and δT of 0.75.

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(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) Naval Postgraduate School Monterey, California 93940		2a. REPORT SECURITY CLASSIFICATION Unclassified	
		2b. GROUP	
3. REPORT TITLE Development of a Test Procedure for Sputtered Gold as a Solid Film Gear Tooth Lubricant			
4. DESCRIPTIVE NOTES (Type of report and, inclusive dates) Thesis for Master of Science Degree, April 1969			
5. AUTHOR(S) (First name, middle initial, last name) Dennis Richard Jones			
6. REPORT DATE April 1969		7a. TOTAL NO. OF PAGES 48	7b. NO. OF REFS 14
8a. CONTRACT OR GRANT NO. b. PROJECT NO. N/A c. d.		9a. ORIGINATOR'S REPORT NUMBER(S) N/A 9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report) N/A	
10. DISTRIBUTION STATEMENT Distribution of this document is unlimited.			
11. SUPPLEMENTARY NOTES		12. SPONSORING MILITARY ACTIVITY Naval Postgraduate School Monterey, California 93940	
13. ABSTRACT Further work on a solid film lubricant testing apparatus designed and built by M. M. Sampsel was carried out. A power supply was constructed and a sputtered gold film applied to various portions of the apparatus. A torsion bridge was installed and calibrated to determine tooth contact forces. Calibration of strain gages installed to determine tooth radial forces was attempted but difficulties were encountered in determining a strain reading for the unstressed condition. Based on this work, recommendations for design changes were made.			

14

KEY WORDS

LINK A

LINK B

LINK C

ROLE

WT

ROLE

WT

ROLE

WT

Sputtered Gold

Solid Film Lubricant

Gear Test Apparatus

